Optimum Design of Damped Dynamic Vibration Absorber – A Simulation Approach

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Abstract: Vibration mitigation is essential in mechanical systems. Reduction in vibration levels improves the life, performance and functionality of machines and equipment. The most effective way of reducing the vibrations is addition of auxiliary mass through stiffness element which is known as Dynamic Vibration Absorber (DVA). It can be with or without damper depending on application requirement. Optimum design of Damped Dynamic Vibration Absorber (DDVA) presents conflicting requirements. Mass ratio (μ), damping ratio (ξ) and tuning ratio (f) needs to be optimally selected to realize satisfactory performance of modified system. Simulation provides better insights in dynamical behavior and gives quantitative understanding about the above dimensionless parameters. Taguchi method parametrically optimizes to reach concrete design decisions. Experimental work involves design of DDVA for parameters derived from simulation. Performance is assessed in absence of absorber system, with undamped absorber and damped absorber system. Effectiveness of DDVA assessed over a speed range over which main system is most likely to operate.

1. Introduction

The *vibration absorber* [1], also called *dynamic vibration absorber (DVA)*, is a mechanical device used to reduce or eliminate unwanted vibration. It consists of auxiliary system attached to the main system that needs to be protected from excessive vibration. Thus the main system and the attached absorber mass constitute a two-degree-of-freedom system; hence the modified system will have two natural frequencies.

The vibration absorber is commonly used in machinery that operates at constant speed, because it is tuned to one particular frequency and is effective only over a narrow band of frequencies. Common applications of the vibration absorber include reciprocating tools, such as sanders, saws, and compactors, large reciprocating internal combustion engines and pumps which run at constant speed (for minimum fuel consumption). In these systems, the vibration absorber helps balance the reciprocating forces. Without a vibration absorber, the unbalanced reciprocating forces may make the device impossible to hold or control.

A spring mass DVA [2] operates over narrow frequency range and suffers from performance deterioration with variation in excitation frequency. Performance robustness can be improved by introducing damping. A Damped Dynamic Vibration Absorber (DDVA) has expectedly much lower response than undamped DVA with broadened operational frequency range.

Design of DDVA possesses formidable manufacturing difficulties. Making a DDVA suitable for wide operating range requires elaborate procedure.

2. MATHEMATICAL MODELING OF DDVA

Addition of auxiliary system to main system renders it potentially a two degree of freedom system. Parametric model of the DDVA is presented in Fig. 1 [2].
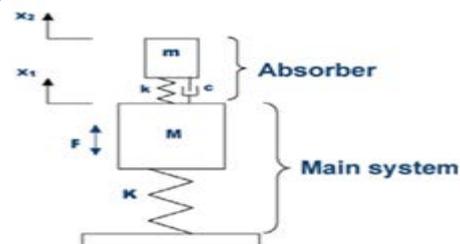

Fig. 1 Two DOF damped DVA model for Simulation

In the Fig. 1, an auxiliary system comprising of a small mass 'm' is attached to main system of mass 'M' so that vibrations of primary mass gets controlled.

M, m = mass of the primary and auxiliary systems in kg respectively.

K, k = spring stiffness of the primary and auxiliary systems in N/m respectively.

F = excitation force in N

x_1, x_2 = displacement of primary and auxiliary systems in meter.

c = damping in N-s/m

The equations of motion for above system are as given below [3],

$$M\ddot{x}_1 + Kx_1 + k(x_1 - x_2) + c(\dot{x}_1 - \dot{x}_2) = F_0 \sin \omega t$$

(01)

$$m\ddot{x}_2 + k(x_2 - x_1) + c(\dot{x}_2 - \dot{x}_1) = 0 \qquad (02)$$

For simplification we convert these equations to dimensionless form by using following symbols:

$\mu = m/M$ = mass ratio = absorber mass / main mass

$\omega_a^2 = k/m$ = natural frequency of absorber

$\omega_m^2 = K/M$ = natural frequency of main system

$f = \dfrac{\omega_a}{\omega_m}$ = tuning ratio (natural frequencies)

$g = \dfrac{\omega}{\omega_m}$ = forced frequency frequency ratio

$x_{st} = \dfrac{P_0}{K}$ = static deflection of system

$c_c = 2m\omega_a$ = 'critical' damping

$\xi = \dfrac{c}{c_c}$ = damping ratio

$$\qquad (03)$$

The main system response x_1 needs to be controlled as given by the Eq. (04),

$$\frac{x_1}{x_{st}} = \sqrt{\frac{\left(2\frac{c}{c_c}g\right)^2 + (g^2 - f^2)^2}{\left(2\frac{c}{c_c}g\right)^2(g^2 - 1 + \mu g^2)^2 + [\mu f^2 g^2 - (g^2 - 1)(g^2 - f^2)]^2}} \quad (04)$$

The equation clearly indicates the parameters to be controlled viz. mass ratio (μ), damping ratio (ξ) and tuning ratio (f). MATLAB program [4] is prepared for plotting frequency response curves using equation (04).Fig. 2 shows response of main system without auxiliary system. Resonance conditions at tuning frequency are clearly evident from it.
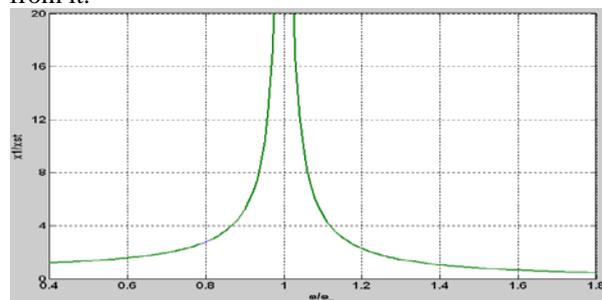

Fig. 2 Response of main system without auxiliary system ($\mu = 0$, $\xi = 0$, f = 0.7 to 0.8)

Response after addition of auxiliary system without damping for mass ratio (μ) = 0.2 and tuning ratio (f) = 0.7 to 0.8 is shown in Fig. 3. The modified system has two natural frequencies one above and one below the original natural frequency. The spread is controlled by the mass ratio (μ).
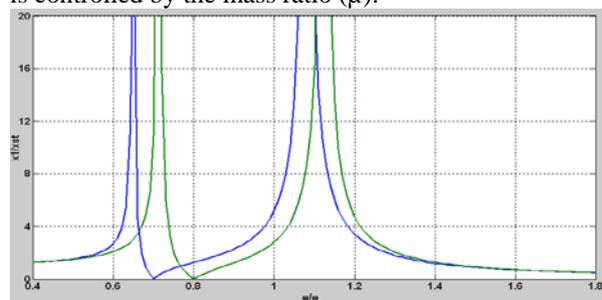

Fig. 3 Response of main system after addition of auxiliary mass (μ = 0.2, ξ = 0, f = 0.7 to 0.8)

The larger absorber mass separates the two natural frequencies wider resulting in safe operating range. But larger absorber mass is highly impractical especially for large machinery. In such cases addition of damping provides better attenuation in a wide frequency range. With increment in the damping, system response further alleviates. (Refer Fig. 4).
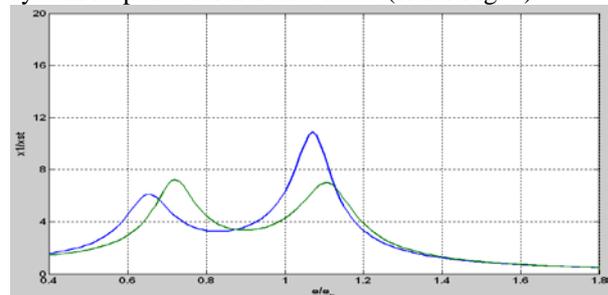

Fig. 4 Response of main system with damping (μ = 0.2, ξ = 0.125, f = 0.7 to 0.8)

Fig. 5 shows the system response after retaining mass ratio (μ) = 0.2, damping ratio (ξ) = 0.25 and tuning ratio (f) = 0.7 to 0.8.
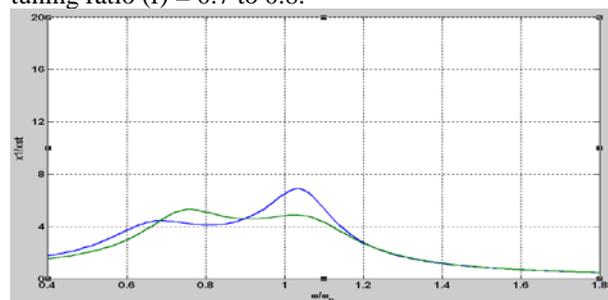

Fig. 5 Response of primary system for μ = 0.2, ξ = 0.25, f = 0.7 to 0.8

In Fig. 6, system response is shown considering mass ratio (μ) = 0.2, damping ratio (ξ) = 0.125 and tuning ratio (f) = 0.9 to 1.0.
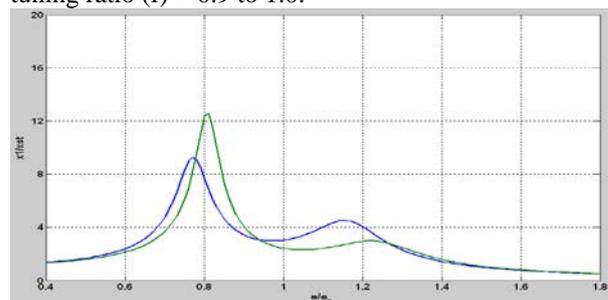

Fig. 6 Response of primary system for μ = 0.2, ξ = 0.125, f = 0.9 to 1.0

Comparison of results from Fig. 5 and Fig. 7 helps in selecting tuning ratio for obtaining optimum results. Merely increase in the damping may reduce the response but it also narrows operating range.

3. SIMULATION OF DDVA IN SIMULINK

Whether we are interested in the behavior of an automotive clutch system, the flutter of an airplane wing or the effect of the monetary supply on the economy, simulation helps to understand wide variety of real world phenomena. Simulation of DDVA is shown in Fig. 2 [5] [6].
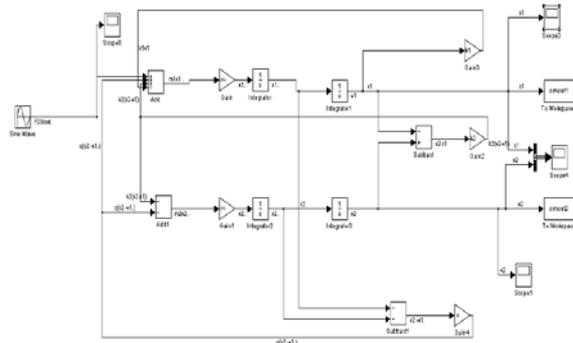

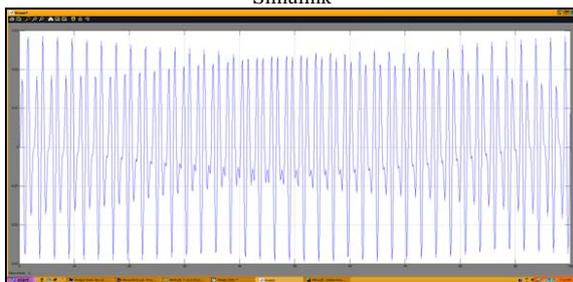Fig. 7 Simulation of two DOF damped DVA model using Simulink

Fig. 8 Simulation results for μ = 0.05, ξ = 0, f = 0.7

Simulation results are obtained for mass ratio (μ) varying from 0.05 to 0.2, damping ratio (ξ) from 0.0 to 0.5 and tuning ratio (f) from 0.7 to 1.0 (Refer Table 1).

TABLE 1
RESPONSE OF PRIMARY AND ABSORBER SYSTEM

Sr. No.	μ (m/M)	f (ω_a/ω_m)	ξ=0.0 x_1	ξ=0.125 x_1	ξ=0.25 x_1	ξ=0.5 x_1
1	0.05	0.7	28.5	30	30	31.5
2	0.1	0.7	27	27	28	32
3	0.15	0.7	26	28	25	30
4	0.2	0.7	23	21.5	24	30
5	0.05	0.8	39	40	40	45
6	0.1	0.8	32	35	37	45
7	0.15	0.8	27.5	30	33	43
8	0.2	0.8	24	27	30	41
9	0.05	0.9	46	50	67	87
10	0.1	0.9	34	40	52	80
11	0.15	0.9	28	32	43	73
12	0.2	0.9	25	28	33	61
13	0.05	1	45	61	95	160
14	0.1	1	32	38	50	80
15	0.15	1	26	30.5	39.5	53
16	0.2	1	22.5	26	30.5	45

4. PARAMETRIC OPTIMIZATION USING TAGUCHI METHOD

Taguchi method [7] is used for parametric optimization of mass ratio (μ), damping ratio (ξ) and tuning ratio (f).

TABLE 2 : MATRIX FOR PARAMETRIC OPTIMIZATION USING TAGUCHI METHOD

Parameters	Values			
Frequency ratio f	0.7	0.8	0.9	1.0
Mass ratio μ	0.05	0.10	0.15	0.20
Damping ratio ξ	0.0	0.125	0.25	0.5

After parametric optimization we get below optimized parameters: f=0.7 to 0.8, μ=0.2 and ξ=0.125.

5. EXPERIMENTAL SET UP

Experimental work involves design of DDVA for parameters derived from simulation. An experimental vibration exciter is used as a primary (main) system for experimental work. a flat plate is given excitation by variable speed drive unit fitted below it. Drive unit consists of two rotating discs driven by variable speed AC motor through a gear train. Both discs have unbalanced mass for creating vibrations which are transferred to flat plate at top of the instrument. The discs rotate in opposite direction so that horizontal component gets nullified and only vertical component of vibrations gets transferred to the plate of the exciter. The experimental set up is as shown in fig. 9.

Fig. 9 Experimental Set Up

The experimentation is carried out for three different scenarios viz. without absorber, with undamped absorber and with damped absorber system. Speed of the system is measured with tachometer and vibrometer is attached to the main system (flat plate) for measuring displacement. The readings of displacement (peak to peak) for various speed ranges are noted and tabulated in Table 3.

TABLE 3 : DISPLACEMENT VALUES IN MICRONS MEASURED DURING EXPERIMENTATION

Sr. No.	RPM	Without Absorber System	RPM	With undamped Absorber System	RPM	With damped Absorber System
1	270	612	260	581	250	517
2	370	690	380	657	400	624
3	470	713	520	683	450	639
4	570	766	610	714	600	671
5	680	804	680	771	700	715

6. RESULTS AND DISCUSSIONS

Mitigation of vibration problems in mechanical systems is a ubiquitous task. Structural modification is least preferred on mechanical systems for which design has frozen. For single speed machines undamped Dynamic Vibration Absorber works satisfactorily. For variable speed machines damped Dynamic Vibration Absorber is the probable solution.

Mathematical modeling and plotting of frequency response curves with MATLAB programming helps in understanding the effect of important absorber parameters mass ratio (μ), damping ratio (ξ) and tuning ratio (f) on performance of mechanical systems. The larger absorber mass separates the two natural frequencies wider and results in safe operating range. But larger absorber mass is highly impractical as results in bulkiness especially for large machinery. In such cases addition of damping provides better attenuation in a wide frequency range. With increment in the damping, system response further alleviates.

In this work most important absorber parameters mass ratio (μ), damping ratio (ξ) and tuning ratio (f) are decided first through simulation. Simulation is run for different 96 combinations of mass ratio (μ), damping ratio (ξ) and tuning ratio (f). For all these combinations response of primary system is noted (Ref. Table 2). Simulations results show that selection of design parameters is a bit complex task. Hence Taguchi method is used to optimize these important parameters of absorber system. It helps to reduce the number of trials required to find the optimum values. After parametric optimization we get below optimized parameters:
F = 0.7 to 0.8, μ = 0.2 and ξ = 0.125

Frequency response curve is plotted for these parameters which depicts the benefits of damped dynamic vibration absorber i.e. better operating speed range and lower response.
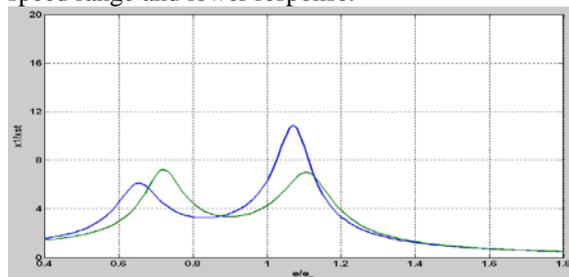

Fig. 10 Response of main system with damping for absorber parameters derived from Taguchi Method (μ = 0.2, ξ = 0.125, f = 0.7 to 0.8)

Experimental model is used to verify designed results. The experimentation is carried for three different scenarios viz. without absorber, with undamped absorber and with damped absorber system. The plot of Displacement Vs RPM is shown in fig. 11. It is clearly evident from the graph that system has better operating speed range and lower response.
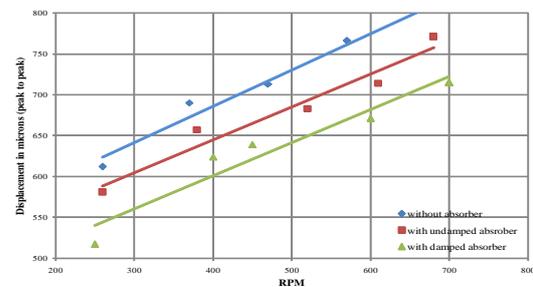

Fig.11 Displacement of primary system at various operating speeds

7. CONCLUSION

Undamped vibration absorber works satisfactorily for single speed machines. Its performance deteriorates with changes in environmental and operating aspects. Damped Dynamic Vibration Absorber eliminates disadvantages of undamped vibration absorber. For variable speed machines damped DVA is the most probable solution. Methodology for finding important design parameters of vibration absorber is proposed by using mathematical modeling, simulation approach and use of Taguchi method. The derived parameters are validated with experimentation. An increase in mass ratio results in diminishing response of main system. But larger absorber mass is highly impractical especially for large machinery. In such cases addition of damping provides better attenuation in a wide frequency range. Increase in frequency ratio also reduces the system response. Similarly, increase in damping ratio results in low response but it also reduces the operating speed range. So choice of design parameters is complex task. By and large, low damping ratio and low tuning ratio can be preferred for low to moderate operating speed range. However, tuning ratio should be high for high operating speed range provided mass ratio is governed by practicalities.

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